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SUMMARY

The performance characteristics of high-speed cylindrical roller bearings can be predicted by using a recently developed computer program called CYBEAN. These characteristics include inner- and outer-race temperatures, component heat generation, cage speed, heat transferred to the lubricant, and bearing fatigue life. Values for these characteristics calculated with CYBEAN were compared with corresponding experimental data obtained previously for a 118-millimeter-bore roller bearing. Operating conditions were radial loads of 2220 to 8900 newtons (500 to 2000 lb), shaft speeds of 10 000 to 25 500 rpm, and total oil flow rates of 0.0038 to 0.0102 cubic meter per minute (1.0 to 2.7 gal/min). The oil inlet temperature was constant at 366 K (200° F).

The cylindrical roller-bearing analysis computer program (CYBEAN) predicted outer-race temperatures and the amount of heat transferred to the lubricant reasonably well. At the higher shaft speeds, the calculated inner-race temperatures were much lower than the corresponding experimental data, unless the effective hot, mounted diametral clearance was set negative by about 0.02 millimeter.

The calculations verified the experimental results, an indication that a bearing can operate with an effective hot, mounted diametral clearance of zero or less. But the calculated values showed that the fatigue life decreases rapidly once all the rollers are in contact with the inner race. The computer program did not predict the cage slip that the experimental study showed with the roller bearing at low shaft speeds.

INTRODUCTION

For the last several years, trends in gas-turbine design have indicated that future aircraft engines may require bearings that can operate reliably at DN values of 3 million or higher (refs. 1 to 3). (The speed parameter DN is the bearing bore in millimeters multiplied by the shaft speed in rpm.) Consequently, there has been a large amount of work done on high-speed bearings in recent years. Successful operation at 3 million DN is reported for ball bearings in references 4 and 5 and for roller bearings in references 3 and 6.

The question of how to design bearings for high-speed applications is increasingly being answered by computer studies (refs. 3 and 7). There are currently several comprehensive computer programs in use that are capable of predicting rolling-bearing operating and performance characteristics. These programs generally accept input data of bearing internal geometry (such as sizes, clearance, and contact angles), bearing

material and lubricant properties, and bearing operating conditions (load, speed, and ambient temperature). The programs then solve several sets of equations that characterize rolling-element bearings. The output produced typically consists of rolling-element loads and Hertz stresses, operating contact angles, component speeds, heat generation, local temperatures, bearing fatigue life, and power loss. In reference 8, data were published that compare computer predictions with actual ball-bearing performance. However, few data have been published that compare computer predictions with actual cylindrical-roller-bearing performance.

As reported in references 9 and 10, a computer program called CYBEAN has recently been developed for analyzing high-speed cylindrical roller bearings. The work reported herein compares the values of inner- and outer-race temperatures, cage speed, and heat transferred to the lubricant, calculated for a 118-millimeter-bore roller bearing by using the computer program CYBEAN, with the corresponding experimental data from reference 6.

BEARING TEST DATA

The experimental data used for comparison purposes in this report were obtained from reference 6. In this reference, a large-bore roller bearing was tested at speeds to 3 million DN, loads to 8900 newtons (2000 lb), and total oil flow rates to 0.0102 cubic meter per minute (2.7 gal/min). Lubrication was provided to the test bearing through axial grooves under the inner race and thence through small radial holes to the rolling elements. The inner ring was also cooled by oil flowing through those axial grooves that did not have any radial holes. About one-half of the total oil introduced to the inner ring was used for cooling only.

The lubricant used was a tetraester, type II oil qualified to the MIL-L-23699 specification. The major properties of the oil are listed in table I. The bearing tester is described in detail in reference 6.

The test bearing was a roller bearing with a 118-millimeter- (4.6457-in.-) bore, a flanged inner ring, and 28 rollers, each 12.65 millimeters (0.4979 in.) in diameter by 14.56 millimeters (0.573 in.) long. Complete specifications are given in table II.

Oil inlet temperature was held constant at 366 K (200° F). Accurate measurement of bearing oil inlet and outlet temperatures allowed determination of the amount of heat transferred to the lubricant at any operating condition. Data were recorded at bearing loads of 2220, 4450, 6670, and 8900 newtons (500, 1000, 1500, and 2000 lb) and at shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The total oil flow rates to the inner ring varied from 0.0019 to 0.0102 cubic meter per minute (0.5 to 2.7 gal/min).

THE COMPUTER PROGRAM

The comprehensive computer program CYBEAN, for analysis of a single cylindrical roller bearing, is completely described in references 9 and 10. This program is capable of calculating the thermal and kinematic performance of high-speed bearings and includes a roller skew prediction for misalined conditions. The calculations include determination of inner- and outer-ring temperatures, oil outlet temperatures, cage speed, and bearing power loss.

Using CYBEAN to predict bearing performance requires as input an estimate of the volume percentage of the bearing cavity that is occupied by the lubricant. The bearing cavity is defined as the space between the inner and outer races that is not occupied by the cage or the rolling elements. The authors of reference 10 recommended that the values used be less than 5 percent. When CYBEAN is also used for a thermal analysis, additional input is required since all the thermal nodes must be defined. The maximum allowable number of nodes is 100. For this investigation 41 nodes were used, including 19 in the lubricant system, as shown in figure 1(a). Later four metal nodes were added (causing the two known-temperature nodes to be renumbered) as shown in figure 1(b).

RESULTS AND DISCUSSION

To effect the direct comparison of predicted and experimental bearing performance, the computer program was generally run at the stated operating conditions of the bearing tested in reference 6. Not all combinations of load, speed, and flow rate were computed, because they are not all necessary for comparison purposes and would require unnecessary computer time. Therefore, the effect of load was calculated at one speed (20 000 rpm), the effect of speed was determined at one load (8900 N (2000 lb)), and the effect of flow rate was observed at one load and two speeds (8900 N (2000 lb) and 20 000 and 25 500 rpm).

To determine how the race temperatures and bearing heat generation vary with the value assumed for the volume percentage of lubricant in the bearing cavity, the computer program was run for several values of this parameter at the 4450-newton (1000-lb), 20 000-rpm condition. The total oil flow rate chosen was 0.0057 cubic meter per minute (1.5 gal/min). The results are shown in figure 2. (In all the figures of this report, the calculated values are always just connected with straight-line segments. Also shown are the corresponding experimental data points.)

The race temperatures (fig. 2(a)) increased with increasing lubricant volume. This would be expected since the fluid drag on the rollers and the cage would increase with the amount of liquid available. Over the full range of 5 percent oil volume, the temperature changes seem to be linear and not too large, about 5 percent at these conditions.

The total heat generated in the bearing (fig. 2(b)) increased with increasing lubricant volume. These changes were also linear, but the total change in heat generation over the volume range was a more significant 50 percent. Since reference 6 includes data on heat transferred to the lubricant (as an indication of the power loss within a bearing), these data were calculated from the computer-predicted oil-outlet temperature and are also shown in figure 2(b). The amount of heat transferred to the oil closely follows the amount of heat generated in the bearing. Over this range of lubricant volume, the amount of heat transferred to the lubricant is about 90 percent of the heat generated in the bearing. Based on the experimental data for this test condition, the range of lubricant volume from 1 to 5 percent is adequate for the outer-race temperature and the heat transferred to the oil. The calculated inner-race temperature remained below the experimental value for the whole volume range.

The computer program was run to determine the effect of bearing load on race temperature and heat generation. Calculations were made for a lubricant flow rate of 0.0057 cubic meter per minute (1.5 gal/min), two lubricant volumes (2 and 3 percent), and a shaft speed of 20 000 rpm. The results, compared with experimental data, are shown in figure 3 for radial loads from 2220 to 8900 newtons (500 to 2000 lb). The predicted race temperatures (fig. 3(a)) increase very slightly over the load range, and the experimental values are practically constant. Although the outer-race temperatures compare favorably, the predicted inner-race temperatures remain about 10 percent lower than the test values. The amount of heat transferred to the oil predicted by using a lubricant volume of 2 percent compares very well with the test data (fig. 3(b)).

The effect of shaft speed was observed by using the computer program for an 8900-newton (2000-lb) load and several shaft speeds. The flow rate was set at 0.0057 cubic meter per minute (1.5 gal/min) and the lubricant volume at 2 percent. The results are shown in figure 4 for shaft speeds from 10 000 to 25 500 rpm (1.2 to 3.0 million DN). The predicted outer-race temperatures (fig. 4(a)) are slightly higher than the experimental data at the lower speeds and slightly lower at the higher speeds. The predicted inner-race temperatures are fairly close to the experimental data at the lower speeds but much lower than the data at the higher speeds. The heat transferred to the oil, however, as predicted by the computer program compared very well with the experimental values over the whole speed range (fig. 4(b)). It is therefore apparent that the calculations for the total bearing heat generation seem to be correct, but that insufficient heat transfer is predicted to the inner race at the higher speeds. At this point it is not clear whether the discrepancy in the inner-race temperatures resulted from using an improper thermal model or from using improper input data.

Since the oil flow rate can have a significant effect on bearing temperature and power loss, the computer program was run over a range of oil flow rates from 0.0038 to 0.0102 cubic meter per minute (1.0 to 2.7 gal/min) for several lubricant volumes.

Calculations were made at 8900-newton (2000-lb) load for both 20 000 and 25 500 rpm. The results are shown in figures 5 and 6. The calculated trends are in the right direction for the values of lubricant volume used; that is, the race temperatures are reduced by increasing the oil flow rate. The calculated outer-race temperatures for 20 000 rpm (fig. 5(a)) are reasonable for lubricant volumes of 2 to 4 percent, but the calculated inner-race temperatures remain low over the entire flow-rate range for these same lubricant volumes. The computed heat transferred to the oil (fig. 5(b)) at 20 000 rpm compares fairly well with the test data over the flow range for these two set lubricant volumes. The comparison at 25 500 rpm (fig. 6) are very similar to those at 20 000 rpm (fig. 5). From these figures, it is not clear how the lubricant volume should vary with flow rate. The outer-race temperatures indicate that the lubricant volume should decrease with flow rate. The heat transferred to the lubricant suggests that the lubricant volume should increase with flow rate. More work needs to be done in this area.

Several calculations were made in an effort to detect any errors in data input or thermal modeling. The program was run at the 8900-newton (2000-lb), 25 500-rpm condition with a flow rate of 0.0102 cubic meter per minute (2.7 gal/min). The convergence criterion (used with an iterative procedure to determine when a solution has been reached as explained in detail in ref. 10) was changed from 0.1 to 0.05 and 0.01, and the calculated temperatures remained the same. A bearing misalinement of 5 minutes was assumed, and the race temperatures changed only 1 kelvin (2 deg F). An additional 300 watts of heat generation was arbitrarily added to nodes 1 and 3 (fig. 1(a)) in case the support-bearing heat generation was not properly accounted for. Although this did raise the inner-race temperatures 3 kelvins (6 deg F), the temperatures of nodes 1 and 3 became unacceptably high, 610 K (638° F). The nodal structure was changed slightly by adding four nodes, two on the shaft and two on the inner-ring adapter (as shown in fig. 1(b)). The resulting calculated temperatures did not change. The value of the heat-transfer coefficient relating the rotating shaft and inner-ring adapter to the air in the rig cavity (e.g., from node 41 to node 21, fig. 1(b)) was changed from 981 to 170 W/m^2 $^{\circ}$C. The inner-race temperature changed 1 degree K. It was concluded that, since none of these parameters had any significant effect on the bearing race temperatures, the inner-race temperatures were mostly affected by the bearing's internally generated heat.

One parameter that could have a large effect on the bearing heat generation is the diametral clearance, that is, the total free movement of the bearing components in a radial direction. Initial calculations with the original thermal nodes (fig. 1(a)) showed only a small change in inner-race temperature from a clearance of 0.12 millimeter (maximum unmounted value) down to 0.001 millimeter. However, reference 6 suggests that a negative clearance exists at 25 500 rpm; so additional calculations were made, with the nodal structure of figure 1(b), for several values of negative clearance. The results are shown in figure 7. The increase in inner-race temperature as the clearance

is lowered below zero is dramatic and closely approaches the experimental value when the clearance is -0.02 millimeter. At this point, and for the first time, all 28 rollers are loaded at the inner ring. At -0.01-millimeter clearance, 13 of the rollers are loaded at the inner race. These calculations suggest that the bearing in reference 6 was very likely operating with a negative clearance at 25 500 rpm.

For comparison purposes, the data of figure 6 were recalculated for a diametral clearance of -0.02 millimeter. These results are shown in figure 8. Both race temperatures (fig. 8(a)) and heat transferred to the lubricant (fig. 8(b)) compare very well with the experimental data.

Although the computer program CYBEAN has the capability of evaluating bearings with out-of-round outer raceways, the program at this point did not predict the effective bearing operating clearance (i.e., the diametral clearance that would exist at operating speed and temperature). Since it became very evident that this was an important parameter for high-speed bearings, subroutines were added to CYBEAN to calculate changes in diametral clearance due to initial fits and due to temperature and high-speed effects. The computer program with these subroutines was then used for further calculations.

CYBEAN was run for several values of bearing unmounted diametral clearance to determine the program-calculated diametral clearance of the mounted bearing at operating speeds and temperatures. This clearance, which does not yet contain the effects of bearing load, will be called the effective hot, mounted clearance. The bearing conditions used were 8900-newton (2000-lb) load, 25 500-rpm shaft speed, 0.0102-cubic-meter-per-minute (2.7-gal/min) lubricant flow rate, and 2 percent lubricant volume. The initial shaft - inner-ring interference was set at 0.0712 millimeter on the diameter. The results are shown in figure 9. With the actual measured value of 0.12 millimeter for cold, unmounted diametral clearance as input, the program predicted about 0.03 millimeter remaining as the effective hot, mounted clearance at operating conditions. To obtain a negative effective clearance, closer to the value of -0.02 millimeter noted previously, required that the input be only 0.09-millimeter initial unmounted clearance. Again, at this point, all 28 rollers are in contact with the inner race. It is interesting to note the large change in fatigue life of this bearing as the clearance gets smaller and the number of rollers in contact with the inner race gets larger. At first, the fatigue life increases and probably becomes a maximum at just that point where all 28 rollers are first in contact. The fatigue life then decreases rapidly as the tighter clearance causes increased stress.

To check program predictions at other conditions, CYBEAN was operated with an input of 0.09-millimeter cold, unmounted diametral clearance for several values of lubricant flow rate. The first calculations were for the 8900-newton (2000-lb), 25 500-rpm condition; the second calculations were for the 8900-newton (2000-lb), 20 000-rpm

condition. The lubricant volume was 2 percent. The results are shown in figures 10 and 11. For the 25 500-rpm shaft speed (fig. 10), the program predicted race temperatures that had the correct trend with lubricant flow rate and were exceptionally close to the experimental values. The calculated heat transferred to the oil (fig. 10(b)) also compared well with the experimental data, although the low-flow-rate value is a little farther off than the others. However, although the predicted trends were correct for 20 000 rpm (fig. 11), the calculated inner-race temperatures were almost 30 kelvins (54 deg F) low. The outer-race temperatures compared fairly well. The calculated heat transferred to the oil (fig. 11(b)) was also fairly close to the data, except at the highest flow rates.

Two final checks were made by using CYBEAN, with the original cold, unmounted diametral clearance of 0.12 millimeter. The first check was with several loads at the low-speed (10 000 rpm) condition. This low-speed condition was chosen because of the large values of cage slip indicated in reference 6. All previous values of calculated cage slip were less than 1 percent at 8900 newtons (2000 lb) and up to 3 percent at 2200 newtons (500 lb). The experimental values at the higher speeds (20 000 and 25 000 rpm) were all less than 2 percent. The flow rate was set at 0.0102 cubic meter per minute (2.7 gal/min) and the lubricant volume at 2 percent. The results are shown in figure 12. Here, the inner-race temperature predictions are close to the experimental values, and the outer-race temperature predictions are about 20 kelvins (36 deg F) higher than the corresponding data. The calculated heat transferred to the oil compares well (fig. 12(b), being slightly higher than the test values. However, although the tests indicated cage slips of over 46 percent for the entire load range, 2220 to 8900 newtons (500 to 2000 lb), the corresponding predicted values were all less than 3 percent. Without this high slip, the experimental temperatures would have been somewhat higher, judging from reference 6 where the bearing temperatures varied inversely with cage slip. The calculated effective hot, mounted diametral clearance was about 0.08 millimeter.

The second check was made by using the program at the high-speed condition (25 500 rpm) with 8900-newton (2000-lb) load and a misalinement angle of 5 minutes to see if the resulting skew would be sufficient to change the predicted race temperatures significantly. This calculation showed the inner-race temperature to be only 8 kelvins higher with skewing and the outer-race temperature to be 3 kelvins higher. The heat transferred to the lubricant was the same. The calculated effective hot, mounted clearance was about 0.03 millimeter. Since the 5-minute misalinement angle is large for this test rig, it can be concluded that misalinement alone would not have been sufficient to cause the experimental inner-race temperatures to be so much higher than the calculated values. Since the effect of misalinement on bearing temperatures was small and the amount of computer time increased by a factor of 10, no further calculations were made with misalinement.

CONCLUDING REMARKS

In general, the CYBEAN computer program, as used in this study, predicted values of outer-race temperature and heat transferred to the oil that compared fairly well with the corresponding experimental data. However, the calculated values of inner-race temperature were usually somewhat low, especially at the higher shaft speeds. The program did not predict the high cage slip experienced in reference 6 at the lower shaft speeds. This is probably the reason that the experimental temperatures were lower than the calculated values for those conditions. Nevertheless, on the basis of absolute temperatures, all calculated values were within 10 percent of the corresponding experimental data and most were within 5 percent. Considering the nature of heat-transfer calculations, this is a reasonably close correlation.

The calculations done for this study show the importance of the effective hot, mounted diametral clearance for useful bearing operation. Care should be taken that the bearing effective diametral clearance remains positive at all operating conditions to assure obtaining a reasonable fatigue life.

As in reference 8 for the computer program SHABERTH, the largest unknown quantity of the input data required for CYBEAN is the volume percentage of lubricant in the bearing cavity. The values chosen for these calculations were in the range recommended in reference 10. From the comparisons presented in this report, it can be concluded that the lubricant volume percentages used are reasonably correct for this program. However, just how these values should vary with oil flow rate and/or shaft speed is still not clear. Further work still needs to be done in this area.

SUMMARY OF RESULTS

The computer program CYBEAN was used to predict inner- and outer-race temperatures, cage speed, and heat transferred to the lubricant for a 118-millimeter-bore cylindrical roller bearing. The results, calculated over a range of operating conditions, were compared with experimental data obtained previously. The bearings were operated at radial loads of 2220, 4450, 6670, and 8900 newtons (500, 1000, 1500, and 2000 lb) and at shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The bearings were lubricated and cooled by flowing oil through and under the inner race at total rates of 0.0038 to 0.0102 cubic meter per minute (1.0 to 2.7 gal/min). The oil inlet temperature was maintained constant at 366 K (200° F). The following results were obtained:

1. The cylindrical roller bearing analysis computer program (CYBEAN) can predict the outer-race temperatures and the amount of heat transferred to the lubricant reasonably well.

2. At the higher shaft speeds, the calculated inner-race temperatures were much lower than the corresponding experimental data, unless the effective hot, mounted diametral clearance was set negative about 0.02 millimeter.

3. A bearing can operate with an effective hot, mounted diametral clearance of zero or less, but the calculated fatigue life decreases rapidly once all of the rollers are in contact with the inner race.

4. The computer program did not predict the high cage slip experimentally obtained with the roller bearing at low shaft speeds.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 13, 1979,
505-04.

REFERENCES

1. Brown, Paul F.: Bearings and Dampers for Advanced Jet Engines. SAE Paper 700318, Apr. 1970.
2. Povinelli, V. P., Jr.: Current Seal Designs and Future Requirements for Turbine Engine Seals and Bearings. J. Aircr., vol. 12, no. 4, Apr. 1975, pp. 266-273.
3. Brown, P. F.; et al.: Mainshaft High Speed Cylindrical Roller Bearings for Gas Turbine Engines. PWA-FR-8615, Pratt & Whitney Aircraft Group, Apr. 1977. (AD-A052351.)
4. Holmes, P. W.: Evaluation of Drilled-Ball Bearings at DN Values to Three Million. I - Variable Oil Flow Tests. NASA CR-2004, 1972.
5. Signer, H.; Bamberger, E. N.; and Zaretsky, E. V.: Parametric Study of the Lubrication of Thrust Loaded 120-mm Bore Ball Bearings to 3 Million DN. J. Lubr. Tech., vol. 96, no. 3, July 1974, pp. 515-524.
6. Schuller, Frederick T.: Operating Characteristics of a Large-Bore Roller Bearing to Speeds of 3×10^6 DN. NASA TP-1413, 1979.
7. Kennedy, Francis E.; and Cheng, Herbert S., eds.: Computer-Aided Design of Bearings and Seals. American Society of Mechanical Engineers, 1976.
8. Coe, Harold H.; and Zaretsky, Erwin V.: Predicted and Experimental Performance of Jet-Lubricated 120-Millimeter-Bore Ball Bearings Operating to 2.5 Million DN. NASA TP-1196, 1978.

9. Kleckner, R. J.; and Pirvics, J.: High Speed Cylindrical Roller Bearing Analysis, SKF Computer Program CYBEAN. Volume I: Analysis. (AL78P022-VOL-1, SKF Industries, Inc.; NASA Contract NAS3-20068.) NASA CR-159460, 1978.

10. Kleckner, R. J.; and Pirvics, J.: High Speed Cylindrical Roller Bearing Analysis. SKF Computer Program CYBEAN. Volume II: Users Manual. (AL78P023-VOL-2, SKF Industries, Inc.; NASA Contract NAS3-20068.) NASA CR-159461, 1978.

TABLE I. - PROPERTIES OF TETRAESTER LUBRICANT[a]

Additives	Antiwear, oxidation inhibitor, and antifoam
Kinematic viscosity, cS, at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Specific heat at 477 K (400° F), J/kg K; Btu/lb °F	2340 (0.54)
Thermal conductivity at 477 K (400° F), J/m sec K; Btu/hr ft °F	0.13 (0.075)
Specific gravity at 477 K (400° F)	0.850

[a]From reference 5.

TABLE II. - ROLLER-BEARING SPECIFICATIONS

Inner race	
Bore diameter, mm (in.)	118 (4.6457)
Raceway diameter, mm (in.)	131.66 (5.1834)
Flange diameter, mm (in.)	137.47 (5.4122)
Total width, mm (in.)	26.92 (1.060)
Groove width, mm (in.)	14.59 (0.5746)
Flange angle, deg	0.6
Outer race	
Outer diameter, mm (in.)	164.49 (6.4760)
Raceway diameter, mm (in.)	157.08 (6.1842)
Total width, mm (in.)	23.9 (0.942)
Rollers	
Diameter, mm (in.)	12.65 (0.4979)
Length, mm (in.):	
Overall	14.56 (0.5733)
Effective	13.04 (0.5133)
Flat	8.40 (0.3307)
Crown radius, mm (in.)	622.3 (24.5)
End radius, mm (in.)	381.0 (15)
Number	28
Cage	
Land diameter, mm (in.)	137.95 (5.4312)
Axial pocket clearance, mm (in.)	0.020 (0.0008)
Tangential pocket clearance, mm (in.)	0.221 (0.0087)
Single rail width, mm (in.)	4.6 (0.18)

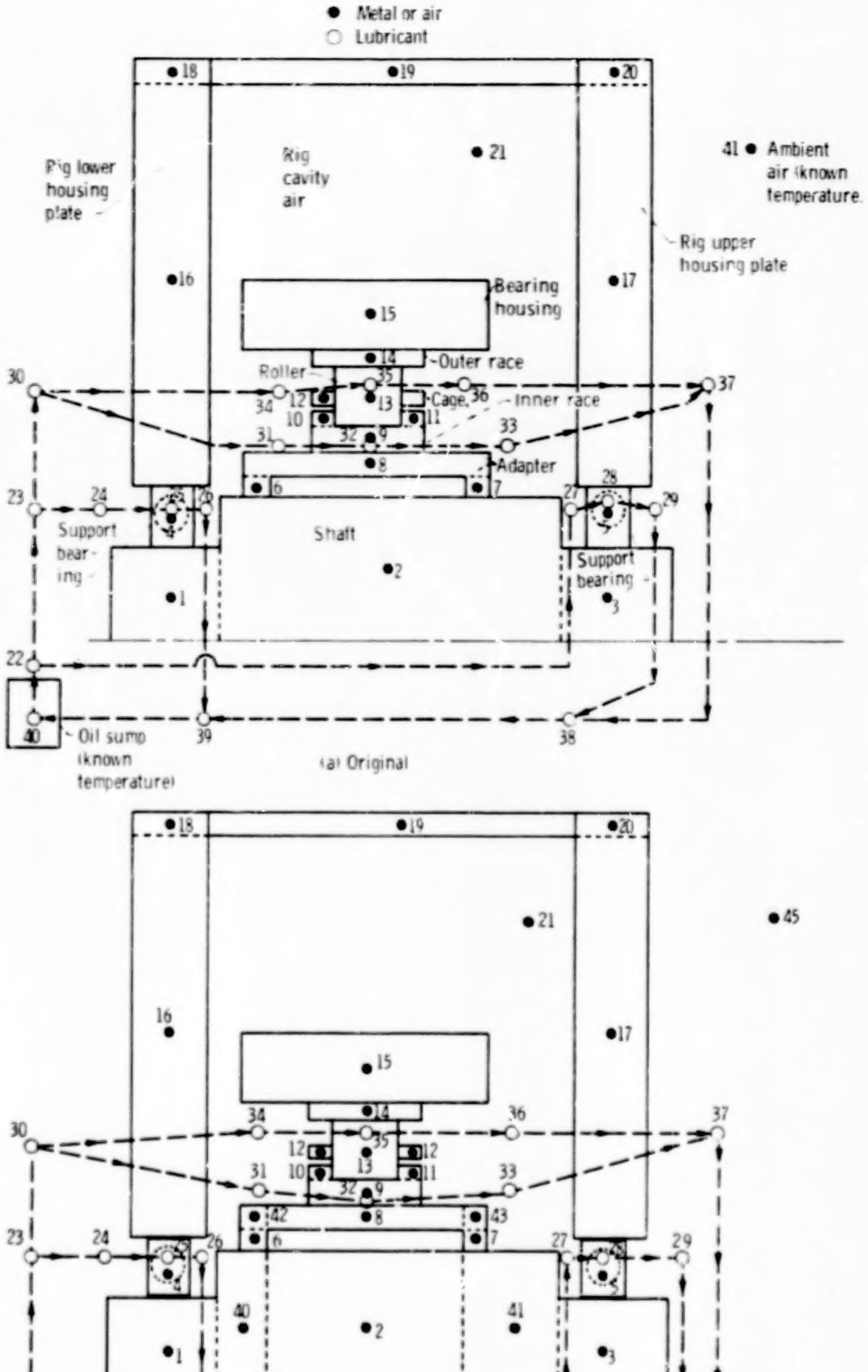
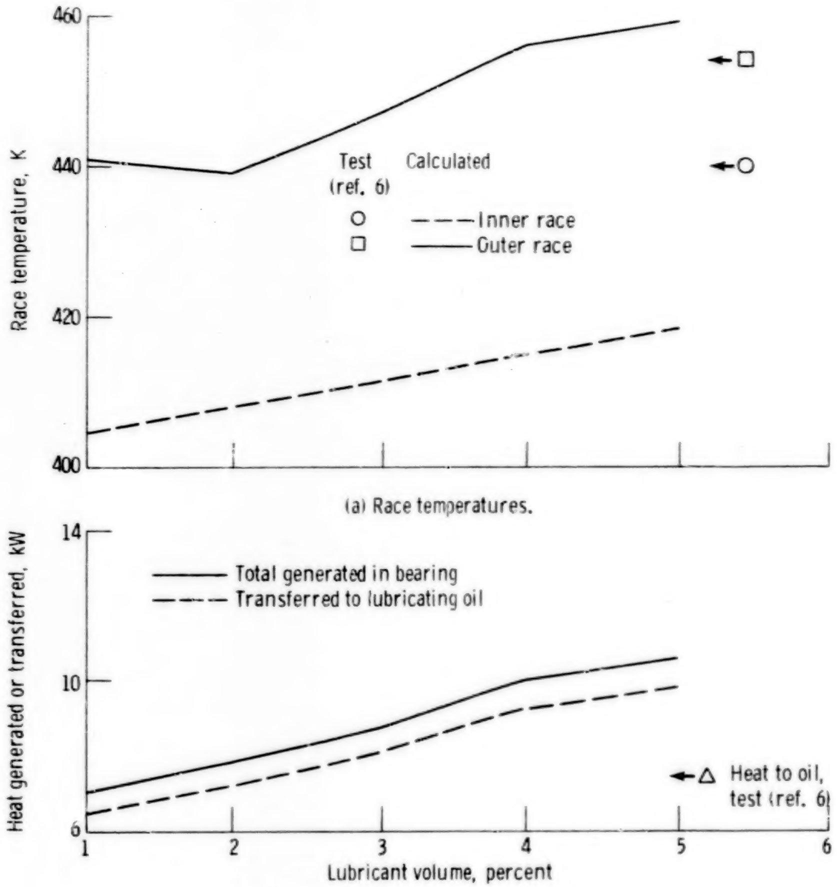
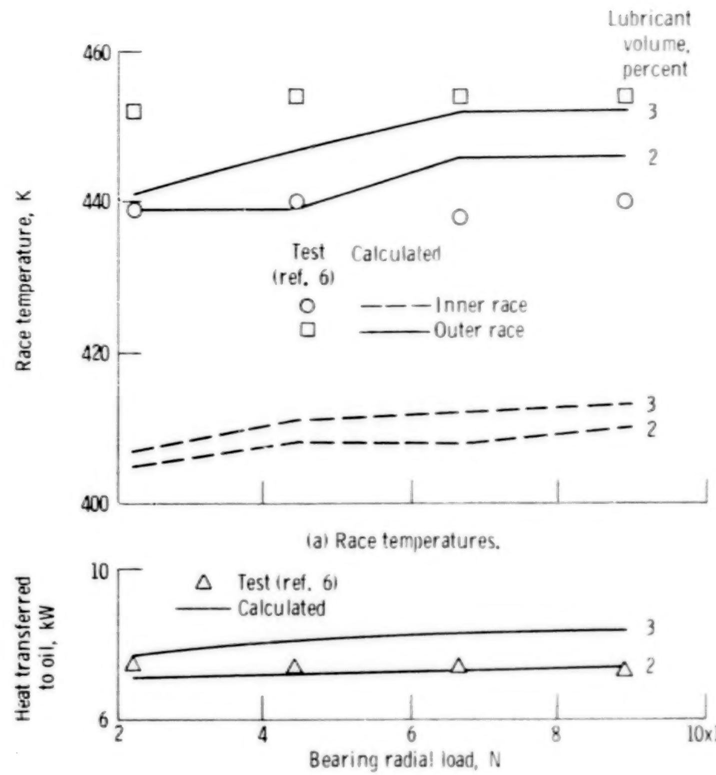


Figure 1. - Nodal system used for thermal routines in CYBEAN.



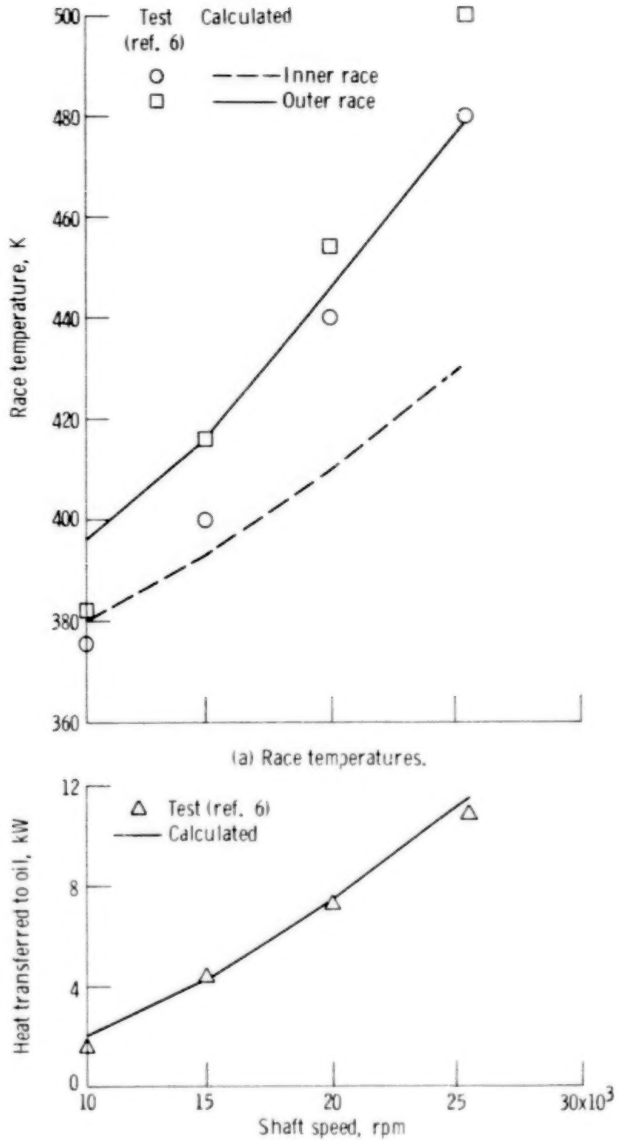
(b) Heat generated in bearing or transferred to lubricant.

Figure 2. - Calculated values of bearing operating characteristics as a function of lubricant volume. (Experimental data are shown for comparison.) Load, 4450 N (1000 lb); shaft speed, 20 000 rpm; lubricant flow rate, 0.0057 m^3/min (1.5 gal/min).



(b) Heat transferred to lubricating oil.

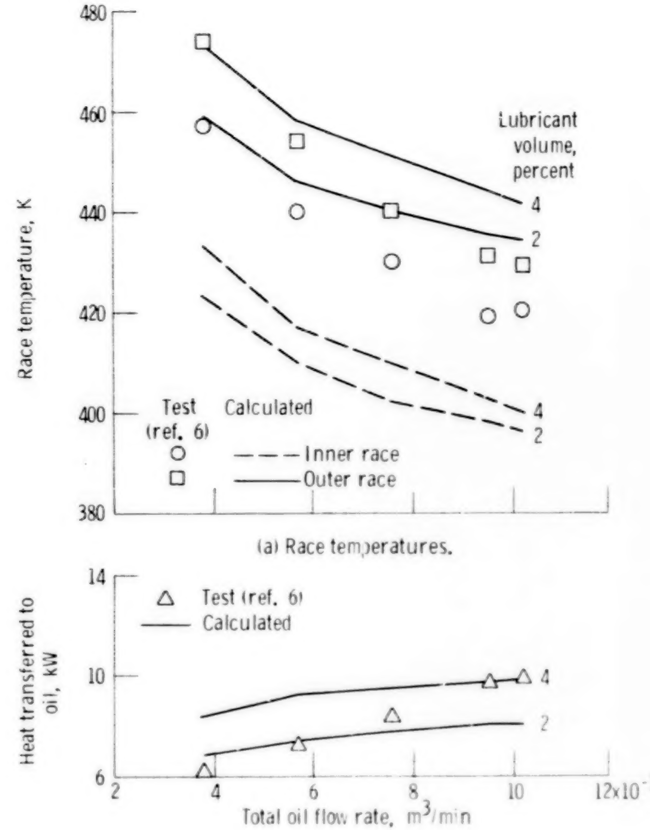
Figure 3. - Calculated and experimental values of bearing operating characteristics as functions of radial load. Shaft speed, 20 000 rpm; lubricant flow rate, 0.0057 m^3/min (1.5 gal/min).



(a) Race temperatures.

(b) Heat transferred to lubricating oil.

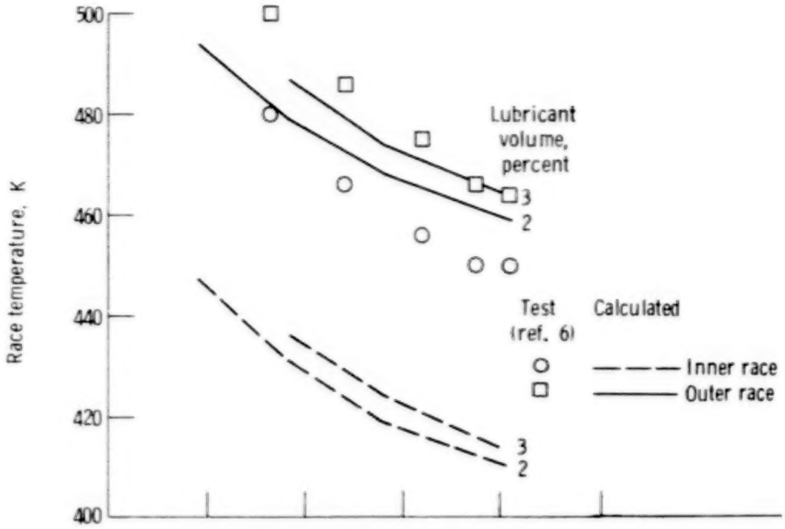
Figure 4. - Calculated and experimental values of bearing operating characteristics as functions of shaft speed. Load, 8900 N (2000 lb); lubricant flow, 0.0057 m^3/min (1.5 gal/min); lubricant volume, 2 percent.



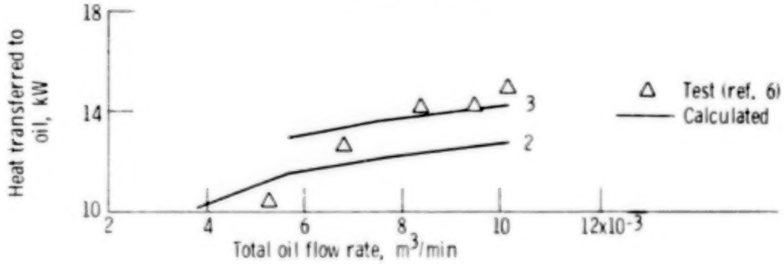
(a) Race temperatures.

(b) Heat transferred to lubricating oil.

Figure 5. - Calculated and experimental values of bearing operating characteristics as functions of total oil flow rate for a shaft speed of 20 000 rpm. Radial load, 8900 N (2000 lb).



(a) Race temperatures.



(b) Heat transferred to lubricating oil.

Figure 6. - Calculated and experimental values of bearing operating characteristics as functions of total oil flow rate for a shaft speed of 25 500 rpm. Radial load, 8900 N (2000 lb).

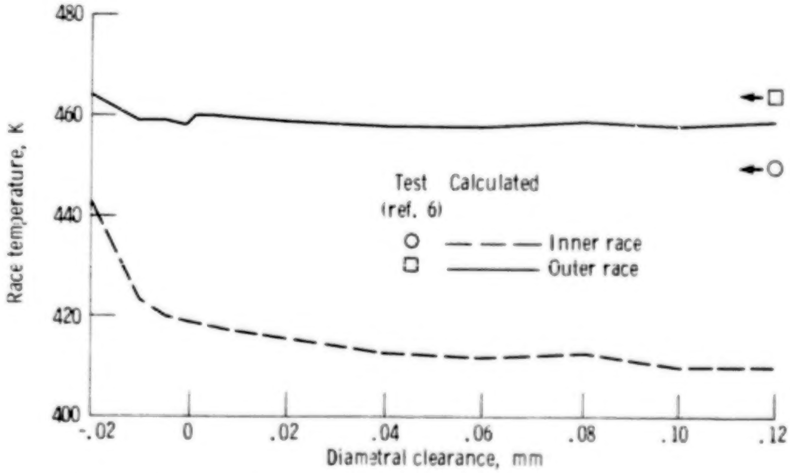
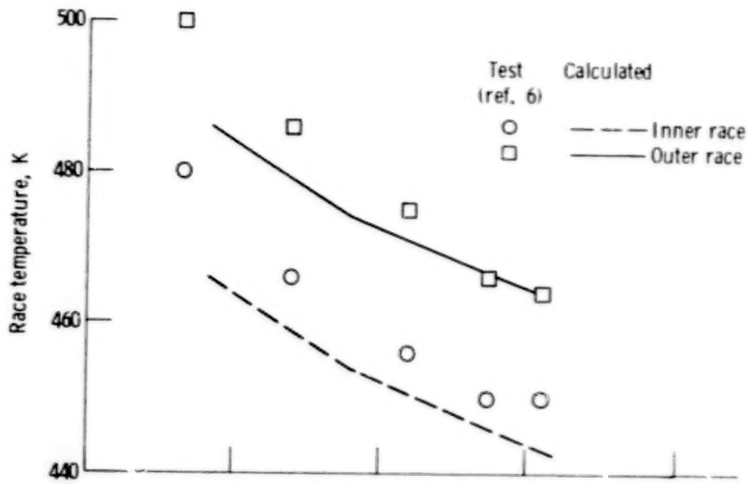
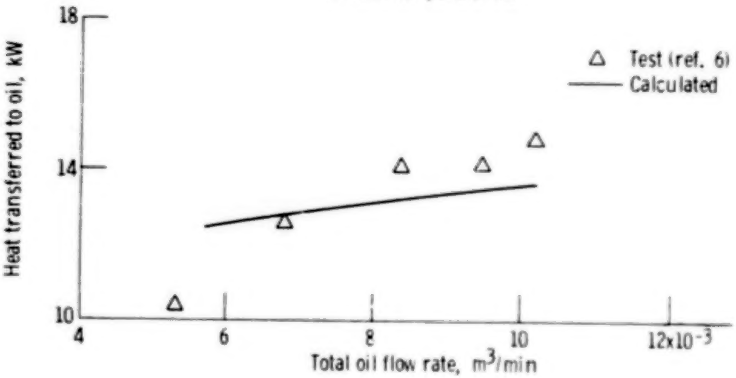


Figure 7. - Calculated race temperatures as functions of diametral clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); total oil flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent. (Test values plotted at maximum possible clearance.)



(a) Race temperatures.



(b) Heat transferred to lubricating oil.

Figure 8. - Calculated and experimental values of bearing operating characteristics with a diametral clearance of -0.02 mm in the computer program. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant volume, 2 percent.

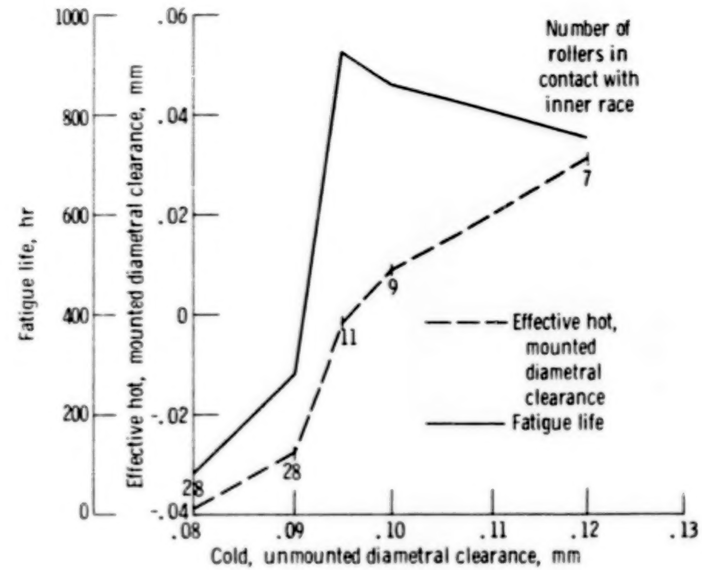
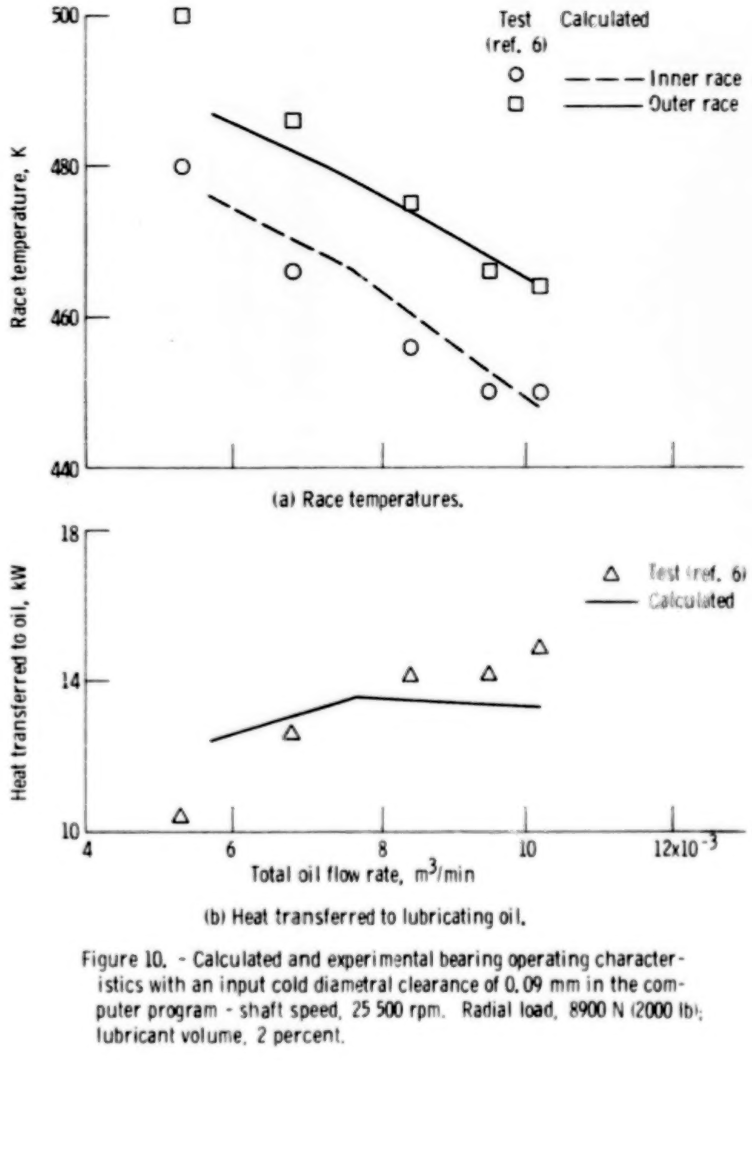
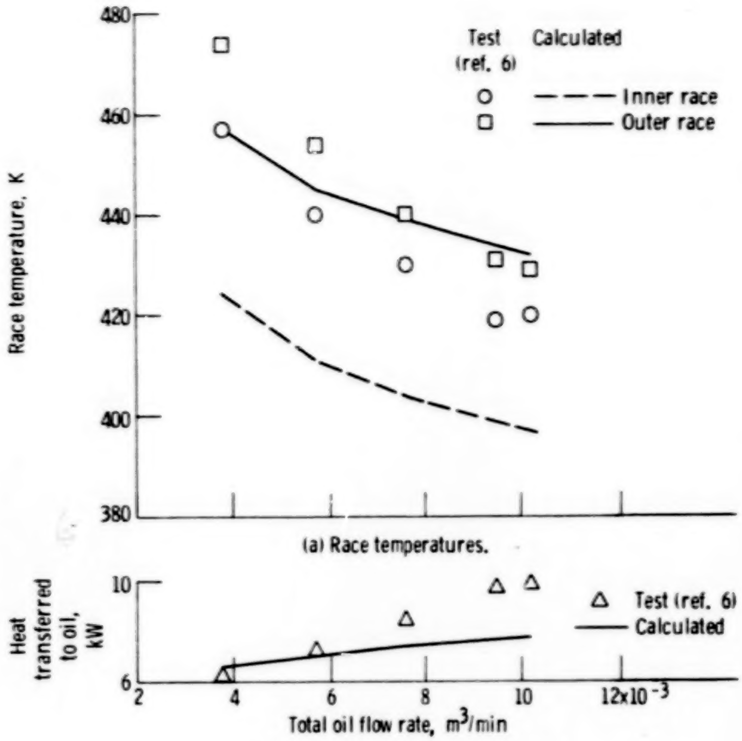


Figure 9. - Calculated values of effective hot, mounted clearance and fatigue life as functions of cold, unmounted clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); oil flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.



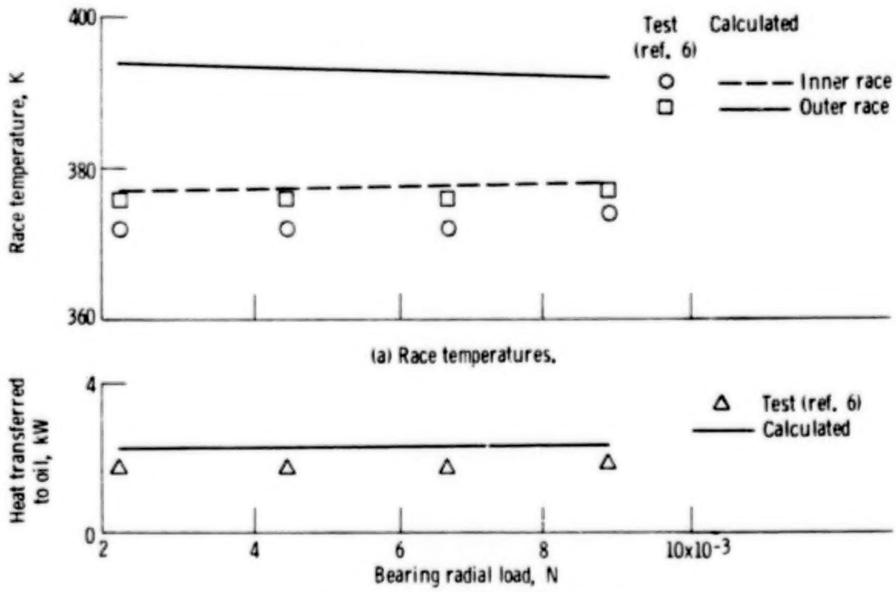
(b) Heat transferred to lubricating oil.

Figure 10. - Calculated and experimental bearing operating characteristics with an input cold diametral clearance of 0.09 mm in the computer program - shaft speed, 25 500 rpm. Radial load, 8900 N (2000 lb); lubricant volume, 2 percent.



(b) Heat transferred to lubricating oil.

Figure 11. - Calculated and experimental bearing operating characteristics with an input cold diametral clearance of 0.09 mm in the computer program - shaft speed, 20 000 rpm. Radial load, 8900 N (2000 lb); lubricant volume, 2 percent.



(b) Heat transferred to lubricating oil.

Figure 12. - Calculated and experimental bearing operating characteristics with an input cold diametral clearance of 0.12 mm in the computer program. Shaft speed, 10 000 rpm; total oil flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

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16. Abstract

Bearing inner- and outer-race temperatures and the amount of heat transferred to the lubricant were calculated by using the computer program CYBEAN. The results obtained were compared with previously reported experimental data for a 118-mm-bore roller bearing that operated at shaft speeds to 25 500 rpm, radial loads to 8900 N (2000 lb), and total lubricant flow rates to 0.0102 m^3/min (2.7 gal/min). The calculated results compared well with the experimental data.

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